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SUBSTITUTES FOR WOODEN BREAKPINS

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SUBSTITUTES FOR WOODEN BREAKPINS

A. H. HOFFMAN[1] AND E. G. McKIBBEN[2]

Including a safety element in the hitch between tractor or team and tillage implement will probably continue to be regarded as good insurance so long as large stones and roots must be encountered. It is almost impossible for the average blacksmith to put a bent plow beam back into correct shape and restore the original degree of stiffness. If the shape is incorrect, the plow will not scour properly. If the rehardening is not properly done, the beam will probably soon bend out of shape again in ordinary use. A good remedy for a bent plow beam is to buy a new one at once. The best remedy and the cheapest is prevention.

The higher plowing speeds now becoming more common greatly heighten the danger of breakage because the forces developed when a heavy object stops suddenly, tend to increase as the square of the speed. Hence the need of safety devices is still present even though there may be fewer obstructing roots and stones than formerly.

Three methods of lessening the danger are in common use. The first is the time-honored procedure of inserting a "weak link" designed to break and release the load if the force becomes too great for safety; the second, in order to avoid harm, cushions the shock by means of one or more compression springs in the hitch, so that the excessive kinetic energy of the moving tractor is transformed into potential energy in the compressed spring; the third method uses a mechanism designed to disconnect automatically any load that reaches a predetermined value. The usual devices representing these methods are respectively the wooden breakpin, the shock-absorbing spring, and the spring-held overload release. This bulletin reports studies of the three methods, conducted principally in order to find a satisfactory substitute for the wooden breakpin.

WOOD VERSUS METAL FOR BREAKPINS

The farmer has long regarded the wooden breakpin as an unsatisfactory safety release element in tractor-drawn implements. Wood is not sufficiently uniform in strength to give adequate protection and is too bulky for the average implement tool box to hold an ample

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supply. Very frequently the last wooden pin breaks and the plowman must either incur a long delay while more pins are being secured or use a bolt or clevis pin from the tool box to make the connection. Usually the substitution of the metal bolt or pin means no safety insurance whatever. If then the plow encounters a large root or rock, a beam will probably be bent; the results will be delay and the expense of a new beam.

Because data on shearing strengths of wood used as breakpins were not available except in average values[3] useless for the purpose and because the need of more satisfactory breakpins was recognized, the tests here reported were undertaken.

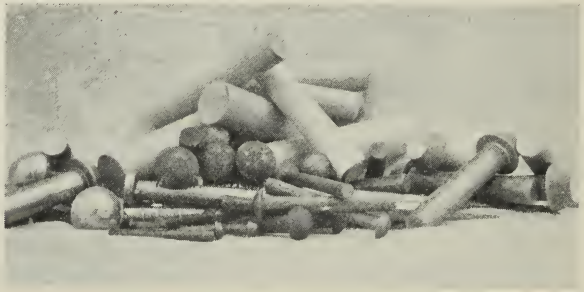

Fig. 1.—The best in breakpins. Maple dowel-pin stock is more uniform than oak. Soft steel rivets and ordinary steel wire nails are better than wood.

The tests corroborate the common experience among operators that the oak pins usually supplied with tractor plows and other heavy tillage implements vary greatly in breaking strength. Often the strongest in a lot of a dozen oak pins, identical in size and make, will carry twice the load that will break the weakest. In one lot in the tests the ratio was 3.8. On the other hand, soft iron or steel rivets are much more uniform in strength, the greatest difference observed between the strongest and the weakest of a size being only about 33 per cent. Steel wire nails are slightly less uniform than rivets. The metal substitutes for breakpins are not only more uniform but less expensive, and so much less bulky that an adequate supply may readily be carried in the implement tool box. Figure 1 shows some of the best kinds of breakpins tested.

Sources of Wooden Pins Tested.—To find out the strength and variations of the ordinary wooden pins, four dozen oak pins were obtained from two implement manufacturers, and broken. These were

[3] Kidder, Frank E., and Thomas Nolan. The architects' and builders' pocket-book. John Wiley and Sons. 16th ed. p. 650–651. 1916.

apparently the ordinary run of the pins supplied to the trade. Some of them, especially the 1 1/16-inch and 7/8-inch sizes, were noticeably widely variant in quality, and the tests showed correspondingly large variation in strength.

The maple pins were made from dowel pin stock secured from two sources, the University Farm store room and the local lumber yard. In both cases the particular stock lengths which were cut up into pins were chosen with the idea of securing maximum variation in quality, some heartwood and some sapwood. Nevertheless, the variation in strength was less than that shown by the oak pins.

The fir (Oregon pine) pins were made by cutting into 3 1/2-inch lengths after ripping to 3/4-inch square and then driving through a die made by drilling a 3/4-inch hole through a bar of 5/8 x 2 1/2-inch iron. The fact that the thirteen fir pins tested were all made from the same 6-foot length of lumber accounts in part for the small range in breaking strength observed for this material.

Sources of Metal Pins.—To find the maximum range of variation, the soft iron rivets were obtained from as many different sources as feasible. There were three lots of 5/16-inch, five lots of 1/4-inch, two lots of 3/16-inch, and one lot each of 1/2-inch, 7/8-inch, and 3/8-inch.

The wire nails were obtained locally or from the store room supply; and each of the three sizes, 20*d*, 40*d*, and 60*d*, was divided into two lots, one of which was tested hard as received, while the other was softened by heating to a good red temperature in a fire of wood, and then leaving to cool as the fire died out, the total annealing time being about four hours. The 30*d*, 50*d*, 70*d*, and 80*d* nails were tested hard as received.

The 1/4-inch bolts were two lots, one of machine bolts and one of carriage bolts, obtained from the store room supply.

The 3/8-inch bolts were a mixed lot of typical toolbox junk, gathered up from various sources, their only points of likeness being that all were nominally iron and 3/8-inch in diameter. The lot contained fifteen machine bolts, five carriage bolts, and six Ford connecting-rod bearing bolts.

All the pins broken were long enough to be in double shear when undergoing test. Maple pins if too short tend to split under stress and therefore to shear more easily. For this reason the maple and fir pins were cut 3 1/2 inches long. The oak pins, though less liable to split, were also of ample length.

Test Apparatus Used.—Four sets of shearing bars were used in the test. One was the regular drawbar head and drawbar of a gang plow (fig. 2). The pin hole in the cast head was ¾-inch in diameter; the punched pin hole in the 2½ x 2-inch drawbar tapered from ¾-inch on the entering side to about $^{11}/_{16}$-inch on the other side. This fact and the resulting compression of the wood probably account for the higher

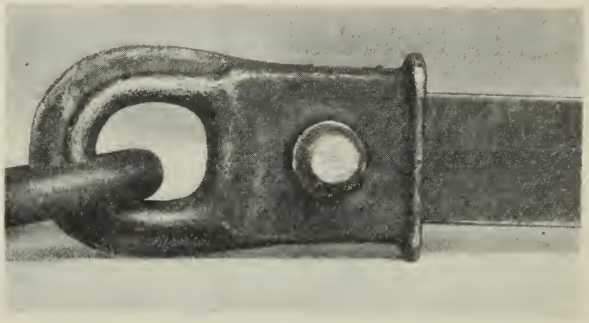

Fig. 2.—A regular cast drawbar head was used for some of the tests.

Fig. 3.—Two sets of three bars were used, each having holes for pins of different diameters. The last test made in the upper set of bars was on a ⅜-inch machine bolt in the ¾-inch hole. The ⅜-inch bars bent badly, and the load went above 15,000 pounds.

breaking strength of the ¾-inch maple pins when forced into this drawbar with a 12-pound hammer. Two sets were of three bars each; one, the upper set in figure 3, had outer bars ⅜-inch by 2-inch and middle bar ½-inch by 1½-inch, of ordinary mild steel, with Brinell hardness numbers 138, and 134–144, respectively, for the side bars and 138 for the middle bar. The lower set had bars ⅝-inch by 2¼-inch of mild steel (Brinell number 154)). Each bar had one end slotted for the clamp bolts. These held the bars together closely, but not tightly. The bars as shown in figure 3 are not spaced. For some

of the tests, spacing washers were used to simulate the clearance often found between cast heads and drawbars. One set of washers spaced the bars 0.041-inch and the other 0.077-inch on each side.

The forty-two ¼-inch soft iron rivets broken in one hole in the thinner set of bars caused only slight deformation of the shearing

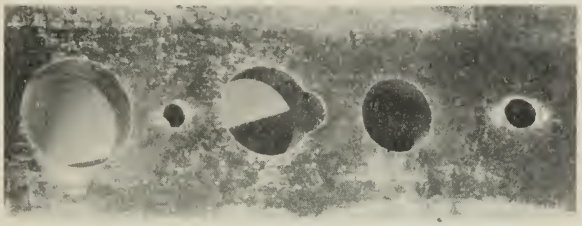

Fig. 4.—The lower bar of the upper set of figure 3. The smallest hole had eleven 3/16-inch iron rivets broken in it; the next (¼-inch) had 42 iron rivets and 10 bolts. The badly marred hole is the ¾-inch bore. The other holes had wooden pins only.

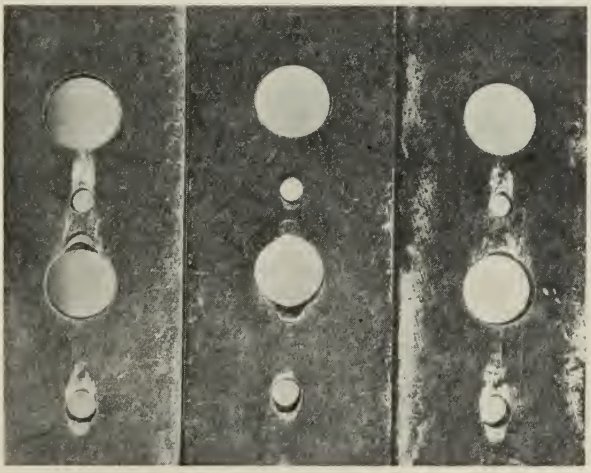

Fig. 5.—The ⅝-inch x 2¼-inch set of bars after the tests. The smallest hole (¼-inch), after twelve 20*d* nails were broken; the next size (5/16-inch) after twelve 40*d* and twelve 60*d* nails were broken; the ¾-inch hole after fourteen ordinary ⅜-inch iron bolts and one Ford connecting-rod bolt were broken. The connecting-rod bolt was broken last.

edges. In the same set of bars a ⅜-inch machine bolt badly bent out one of the ⅜-inch bars and so blunted the shearing edges of the ¾-inch hole in which it was placed (fig. 4), that the load went above 15,000 pounds before the bolt failed. Possibly no bending would have resulted if the ⅜-inch bolt had been in a hole of its own proper size. In the slightly harder ⅝-inch x 2¼-inch bars (fig. 5), the wire nails,

both the annealed and the soft, and fourteen of the ⅜-inch bolts were broken without very much deformation of the shearing edges. The first of the alloy steel connecting rod bolts broken caused in the ⅝-inch bars, however, a very decided deformation, not only of the edges of the ¾-inch hole but for about half the thickness of the outer bars and almost entirely through the middle bar.

The fourth set of shearing bars (fig. 6) was oxyacetylene welded to a form somewhat like the drawbar head of figure 2. It was made of material taken from one ⅝-inch x 2½-inch mild steel bar. The

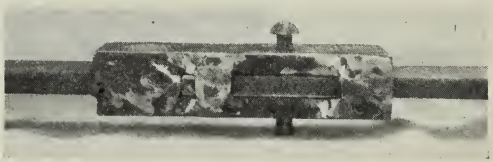

Fig. 6.—A set of shearing bars made up in a form similar to that of figure 2 by oxy-welding ⅝-inch x 2½-inch mild steel bars. This type is less convenient than the bolted for removing the broken pieces of metal pins.

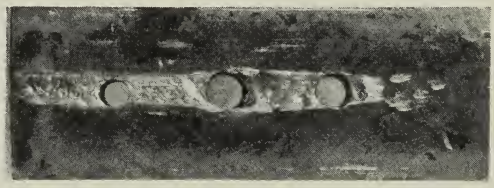

Fig. 7.—The middle bar of the set shown in figure 6. Condition after fifty-nine rivets ⅜ to ½-inch diameter and eight No. 80*d* wire nails had been broken. Although the steel was too soft and flaky for this purpose, the average breaking load was increased only about 12 per cent over that for ordinary hard bars.

hardness was found uniform but somewhat low (Brinell number 121); the steel seemed, besides, to be flaky, and tore off in patches. This set was badly marred in the tests. Figure 7 shows how the surface of the middle bar was torn in the breaking of nineteen ⅜-inch, nine 7/16-inch, twenty-one ½-inch soft steel rivets, and of eight No. 80*d* steel were nails. Some marring resulted when the first pin (a rivet) was broken, but more damage was done by the nails. The breaking loads for the pins broken in this soft drawbar averaged about 12 per cent higher than for other pins of the same lots broken in the lower set shown in figure 3.

Besides a tractor drawing a subsoiler, two testing machines were used to break the pins. One was a Riehle 30,000-pound tension-compression-cross-bending machine; the other was a Riehle 200,000-pound tension-compression machine.

Results of the Tests.—The results of the tests made and the typical variations are shown for wood in figure 8; for metal in figure 9; and for all lots, maxima, minima, and averages are shown in figure 10.

The wooden pins had evidently a much less uniform breaking strength than the metal pins and were weakened by the spacing of

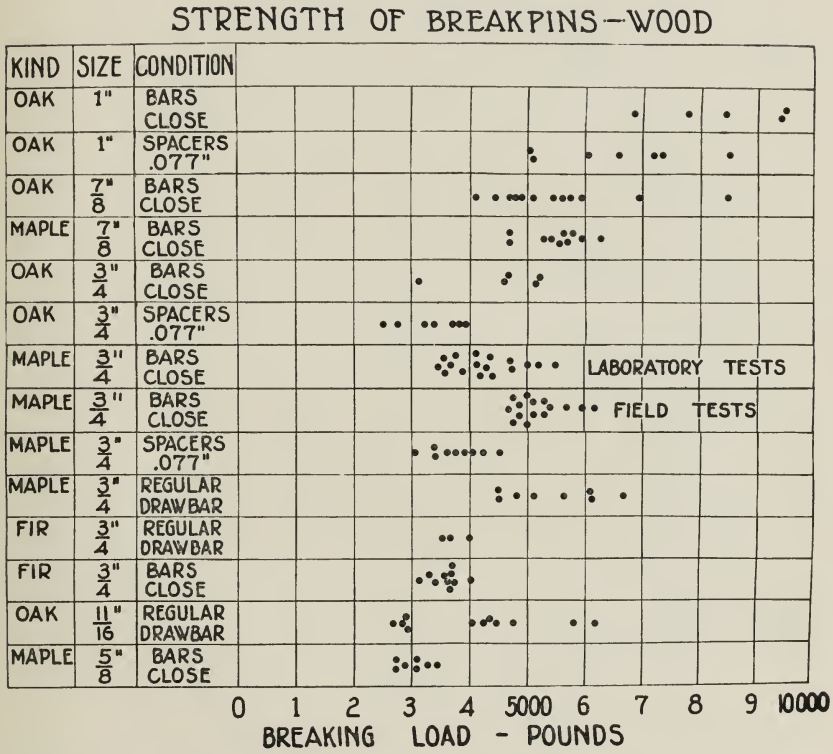

Fig. 8.—Each dot shows the result of a single test. The smaller sizes of oak pins varied most and were most affected by spacing of the bars. Maple pins are more uniform than oak.

the bars more markedly than was the case with the metal pins. The pins of smaller diameter were more affected by spacing than those of larger diameter; spacing seemed, furthermore, to affect the ¾-inch oak pins more than the maple pins of the same size.

A 12-pound hammer was used to drive ¾-inch maple pins into the tapering punched hole in the bar in the regular drawbar head used for some of the tests. Obviously this compression of the the ¾-inch maple pins to $^{11}/_{16}$-inch diameter markedly increased their

breaking strength (fig. 10). These pins were such a tight drive-fit that it was found impossible to get them in without battering them to pieces when a 1¼-pound hammer was used.

The rivets and nails broken show remarkably close uniformity in spite of the deliberate effort made to secure the widest range of

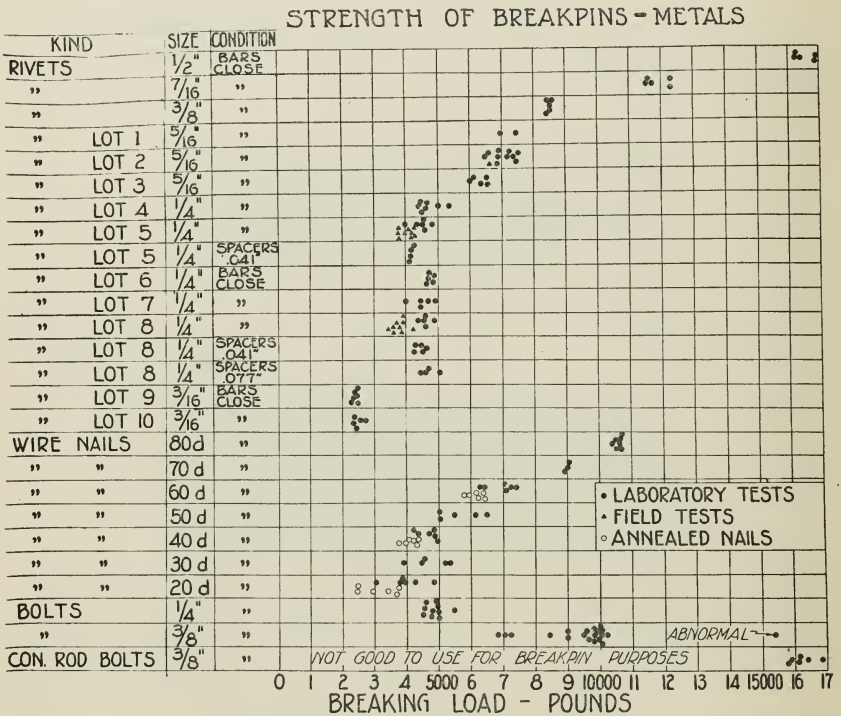
STRENGTH OF BREAKPINS - METALS
KIND
SIZE
CONDITION
RIVETS
1/2" BARS CLOSE
" 7/16 "
" 3/8 "
" LOT 1 5/16 "
" LOT 2 5/16 "
" LOT 3 5/16 "
" LOT 4 1/4" "
" LOT 5 1/4" "
" LOT 5 1/4" SPACERS .041"
" LOT 6 1/4" BARS CLOSE
" LOT 7 1/4" "
" LOT 8 1/4" "
" LOT 8 1/4" SPACERS .041"
" LOT 8 1/4" SPACERS .077"
" LOT 9 3/16" BARS CLOSE
" LOT 10 3/16" "
WIRE NAILS 80d "
" " 70 d "
" " 60 d "
" " 50 d "
" " 40 d "
" " 30 d "
" " 20 d "
BOLTS 1/4" "
" 3/8" "
CON. ROD BOLTS 3/8" "
• LABORATORY TESTS
• FIELD TESTS
○ ANNEALED NAILS
ABNORMAL
NOT GOOD TO USE FOR BREAKPIN PURPOSES
0 1 2 3 4 5000 6 7 8 9 10000 11 12 13 14 15000 16 17
BREAKING LOAD - POUNDS

Fig. 9.—The metal pins were in general more uniform in strength than the wooden pins. Two exceptions were found in testing ⅜-inch junk bolts. Alloy steel connecting-rod bearing bolts went up to nearly twice the average. The other abnormally high load was caused by one set of shearing bars bending because it was too light.

ordinary variation by buying several lots of each size from different sources. Only the miscellaneous lot of ⅜-inch junk bolts shows great difference between the strongest and the weakest. The reasons for this divergence are: (1) Some were carriage bolts with threads rolled up to ⅜-inch from stock of slightly smaller size; (2) some were machine bolts with threads cut on full-size stock; (3) some were special alloy steel of great strength instead of common soft low-carbon steel.

The purpose in testing these junk bolts was not to see whether they would make desirable substitutes for wooden breakpins, but rather to show the danger of using any sort of old bolt for such purposes. Ordinary machine bolts purchased in the market may safely be expected to break at loads very similar to those for soft steel rivets of the same diameter. Similarly common carriage bolts, because

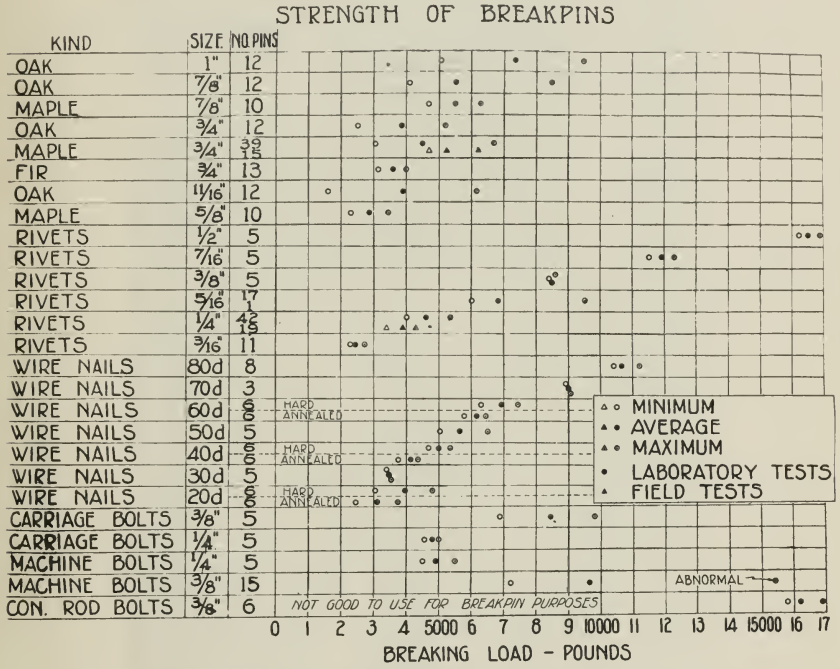

Fig. 10.—Averages, maxima, and minima for all the lots of pins tested. Equivalent strengths may be obtained by comparing averages for metals with maxima for wood. See also table 1.

undersize, will break at about ten per cent smaller load. Bolts salvaged from machines, especially from automobiles, trucks, tractors, and other high class equipment, are never safe for breakpin purposes.

Equivalent Pins of Wood and Metal.—Figure 10 indicates in general what size of rivet, nail, or soft iron bolt may be substituted for an oak pin of a given size. Table 1 gives the same but more accurately. In a trial of ¼-inch rivets substituted for ¾-inch oak pins in a tractor-subsoiler outfit (fig. 11), the rivets released at somewhat lower loads than were found in the laboratory by means of the testing machine. Because of the fact just mentioned, because of the uniformity of strength of the metals, and because only the strongest of wooden pins

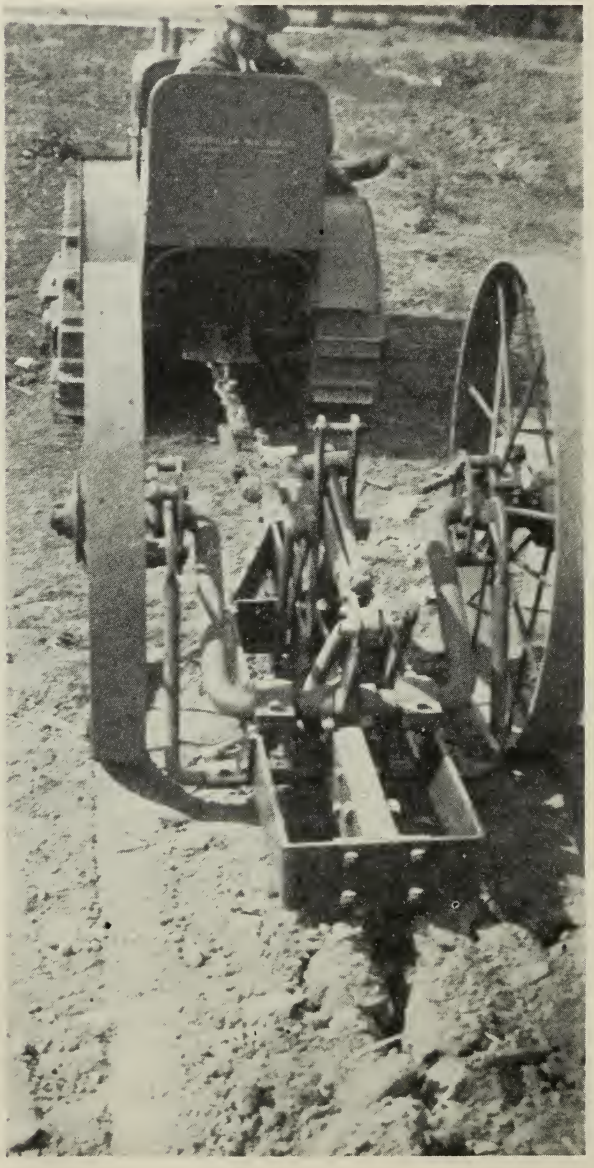

Fig. 11.—A tractor and a subsoiler used to test breakpins. One-fourth-inch rivets, substituted for ¾-inch oak pins, released at average loads slightly lower than in the laboratory tests.

are in use for any appreciable length of time, it was deemed best in making up the table to use the maximum breaking load for the oak pin of a given size and the average or minimum value in finding the size of the metal equivalent.

Adapting Machines for Metal Breakpins.—Most drawbar heads made for wooden breakpins give ample room to drill another hole for the equivalent size of metal pin. It is better to drill the new hole rather than to use metal in the hole designed for wood, because in

TABLE 1

Metal Equivalents for Wooden Breakpins

*Breaking load, pounds	Diameter of pin (inches), or number of nail					
	Wood	Soft steel rivets	Common wire nails	Wire nails annealed	Carriage bolts, rolled thread	Common machine bolts
16,500		1/2				
12,000		7/16				
10,700			80*d*			
9,700						3/8
9,000	1, oak		70*d*			
8,000	7/8, oak				3/8	
7,000		5/16	60*d*			
6,500				60*d*		
5,500	3/4, maple		50*d*			
5,000	3/4, oak		40*d*			1/4
4,800	11/16, oak				1/4	
4,500		1/4				
4,200				40*d*		
4,000	3/4, fir		30*d*			
3,200	5/8, maple		20*d*	30*d*		
2,500		3/16				

* All pins in double shear. The loads for wood are for the strongest pins, for metals the average.

holes of appropriate size the metal pins are more certain to break at the right overload. In the larger holes they might tend to bend light bars and not release in time to protect the machinery. If the 3/16-inch size rivet or a nail smaller than No. 20*d* is needed, as, for example, in cultivators for corn and other row crops, the hole in the drawbars may well be drilled 1/4-inch diameter. If it is drilled to exact size, removal of the broken pieces of small rivets is sometimes found difficult. The rivet or nail chosen should be long enough to extend through, so that the point may be bent over, to prevent its being lost out. For the 1/4-inch size, rivets used for fastening the sickle straps in combines are good. They are 2 1/4-inches long and may be obtained at almost every hardware store in California.

Hardness tests made on several plow drawbar heads designed for wooden breakpins showed Brinell numbers ranging from 176 to 418 for the straight bars and 126 to 137 for the malleable cast heads. These are hard enough to stand up satisfactorily if holes of proper size are drilled and soft steel rivets used as breakpins. While implement manufacturers will undoubtedly use steels of close texture and sufficient hardness (Brinell number above 150) to prevent excessive marring of the edges of the holes, yet, if occasionally a soft drawbar is marred, the increase of the load necessary to break is so slight under even the worst conditions that the implement is safe if protected by a metal breakpin of the proper size.

DRAWBAR SPRINGS[4]

A drawbar spring is a heavy spring, usually of the compression type, placed in the drawbar of a tractor as a shock absorber, to cushion the jerk when the tractor starts a heavy load or abruptly quickens its speed, or when the load resistance is suddenly increased. By preventing the development of enormously large shock forces, it tends to reduce wear and breakage and so prolong the useful life of both tractor and implement.

The possibilities of the drawbar spring are worthy of careful consideration by anyone interested in the use of mechanical power for farm traction purposes. This is particularly true because of the present tendency to increase the speed of tillage operations, and because of the reluctance of the average operator to use any type of complete overload release not absolutely necessitated by soil conditions. It is rather common to replace wooden breakpins with bolts and either to remove or to clamp spring overload release hitches. While the drawbar spring is not a cure-all, and under certain conditions can not replace the breakpin or other methods of complete release, there are times when it can do so; in most cases it will reduce the frequency of breakpin replacement and increase the life of both tractor and implement.

The drawbar spring has the following advantages:

1. Its action is uniform and dependable.

2. It reduces the force applied to the implement and to the tractor, in consequence of a suddenly increased speed or load resistance.

3. It requires, in comparison with breakpins and special release hitches, practically no attention or replacement.

[4] For a full technical discussion see: McKibben, E. G. Substitutes for breakpins—drawbar springs. Agr. Engineering 9:166–170. 1928.

The following are some of the disadvantages and short-comings of the drawbar spring:

1. It adds slightly to the first cost.

2. Entire dependence upon it for overload protection is probably not practicable where tillage implements are used at high speeds.

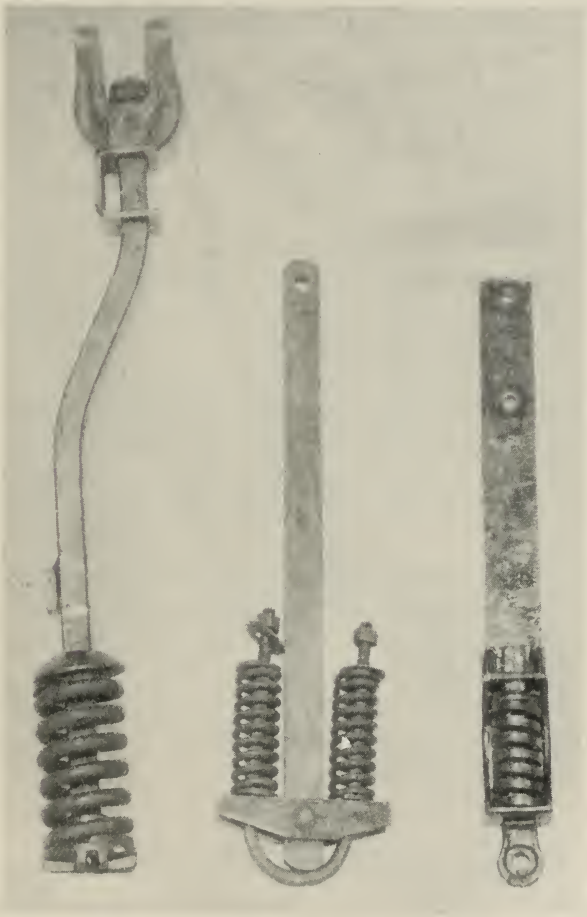

Fig. 12.—Three typical drawbar springs.

3. It does not protect a light implement which is being pulled by a large tractor where the drawbar pull due to traction alone is greater than the safe load for the implement.

From the standpoint of overload protection a drawbar spring is properly designed if it is long enough and strong enough so that when the implement meets a solid obstruction, the inertia force may be overcome and the tractor stopped before any part of either imple-

ment or tractor is permanently bent or broken, and before the spring is completely compressed. Usually a breakpin or some other type of overload release should be used in the hitch bar of the implement even though the tractor is equipped with a drawbar spring.

A drawbar spring designed of proper length and strength for a tractor and its usual implements, will afford little or no protection for much lighter implements. In fact, it is usually not practical under such circumstances to attempt complete overload protection by means of a simple drawbar spring. A breakpin or some other type of overload release should be used.

The inertia force developed when a tractor or other heavy body stops suddenly, tends to vary directly with the weight of the tractor and with the square of the speed. Thus if two tractors, one weighing 1000 pounds and the other 4000, are moving at the same speed, the larger will strike with four times the energy and force of the smaller if both are brought to rest in the same distance. If two tractors of the same weight travel, one at one mile per hour and the other at four, and if both are stopped suddenly under the same conditions, the faster machine will jerk with sixteen times the force of the slower one.

The drawbar spring reduces the force of shocks by increasing the time and distance in which the tractor brings the implement up to speed or in which it comes to rest when the implement encounters a solid obstruction.

Figure 12 shows three typical drawbar springs. Tests showed them all to be too short and weak to afford the maximum of protection, though the hitch shown at the left was found to approach correct design more nearly than the other two. Each of these springs would be completely compressed before the tractor of which it is a part would be stopped and its engine stalled, if the implement to which it is attached encountered a solid obstruction.

SPRING OVERLOAD RELEASE HITCHES[5]

As an effective method of protecting implements against excessive stresses resulting from sudden stopping of the implement by a relatively solid obstruction, the spring overload release hitch deserves a wider consideration by designers and operators of farm tractors and tractor-drawn implements. A properly designed spring overload release hitch may combine most of the advantages of both the breakpin and the simple shock-absorbing drawbar spring, without the more serious disadvantages of either.

[5] For a full technical discussion see: McKibben, E. G. Substitutes for breakpins—spring overload release hitches. Agr. Engineering 9:215–217. 1928.

The following are some of the advantages obtained by the use of a properly designed and well built spring overload release hitch:

1. Protection of the implement against excessive stresses under all conditions of load and speed.

2. Relatively infrequent replacement of parts. (The breakpin must be replaced each time the load is released.)

3. Rather accurate adjustability over a relatively wide range of loads.

4. An appreciable elongation of the hitch before release, which absorbs under some conditions enough of the kinetic energy of the tractor to prevent the necessity of releasing the hitch.

Like all others this type of overload protection has certain disadvantages, as follows:

1. The first cost is usually higher than that of either a simple shock-absorbing drawbar spring or provision for the use of a breakpin.

2. If it is to give satisfactory service it will have to be adjusted occasionally.

3. The inconvenience of rehitching is a disadvantage in comparison with the simple drawbar spring (provided the conditions are such that the situation can be satisfactorily handled by the latter).

Desirable Characteristics in Spring Overload Release Hitches.—The spring overload release hitch should have as many as possible of the following characteristics:

1. Uniformity of pull required for release at any given adjustment.

2. Pull, required for release, proportional to adjustment.

3. Minimum wear of tripping mechanism; that is, of such design that the trip mechanism is subject to the smallest practicable forces and made of high quality materials.

4. Minimum effect of a given amount of wear on the pull required for release at any given adjustment.

5. Adjustability for correcting effect of wear.

6. Convenient, inexpensive replaceability of wearing parts.

7. Ease and convenience in rehitching after release.

8. Positive hitch, so that the implement may be backed and otherwise maneuvered without possibility of its being released unless it is subjected to the predetermined overload.

9. Method of locking against release, to be used to prevent the possibility of a runaway when a relatively free-rolling load is being pulled on a steep grade.

A Study of Three Hitches.—In order to get a better idea of the possibilities and limitations of the spring overload release hitches available at the present time, a brief study was made of three hitches,

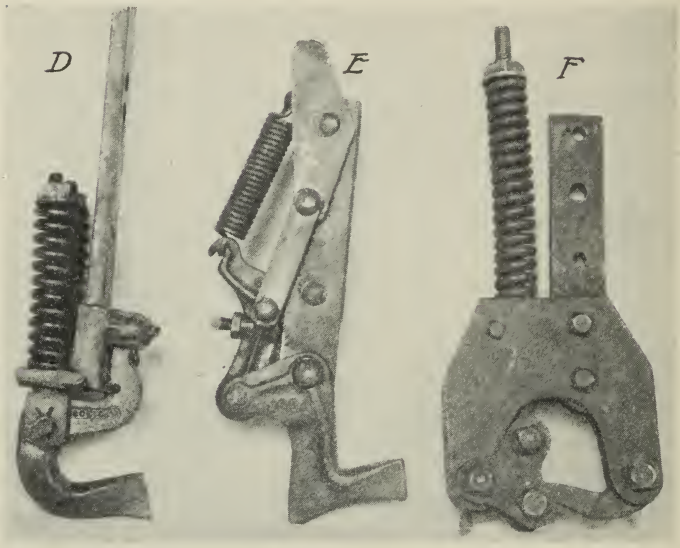

Fig. 13.—Three typical spring overload release hitches.

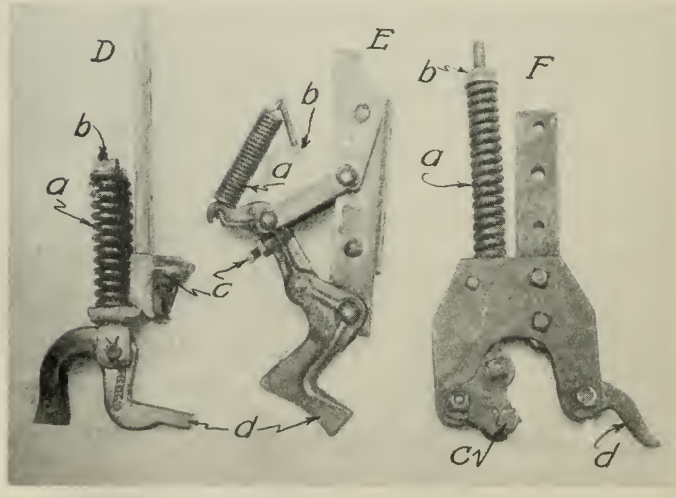

Fig. 14.—The same spring overload release hitches illustrated in figure 13, but showing the hitches in released position. Spring *a* of hitch *E* was disconnected to show the hitch released. Normally this mechanism returns automatically to the position shown in figure 13 as soon as the load has been released.

obtained from three representative implement manufacturers. They are, as shown in figures 13 and 14, of decidedly different types. The results reported below were obtained by use of a recording shock dynamometer between the hitch being studied and the implement.

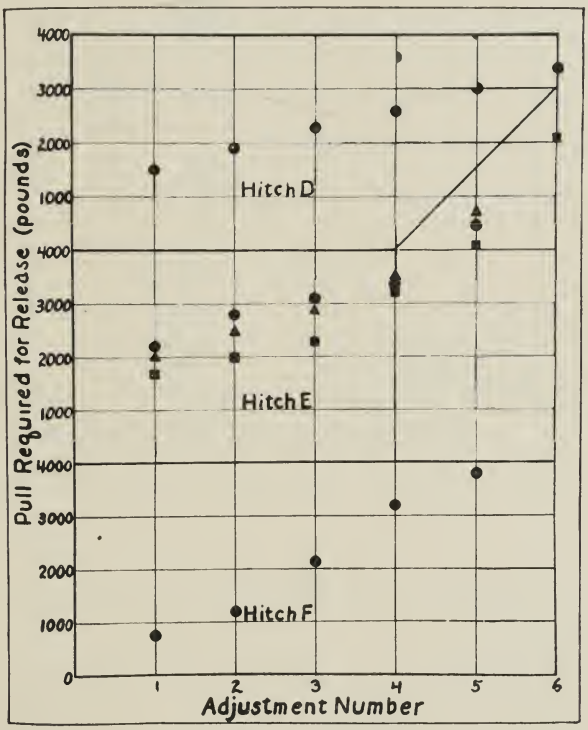

Fig. 15.—Effect of uniform changes in adjustment upon the pull required for release. In the series of tests of hitch *E* shown as round dots, nut *b* (note figure 14) was adjusted to give spring *a* a tension of 150 pounds at adjustment No. 1 of capscrew *c*. Nut *b* was left at this adjustment during the series. In the series of tests of hitch *E* shown as triangles, nut *b* was adjusted to give spring *a* a tension of 150 pounds for each adjustment of capscrew *c*. Likewise in the series shown as squares, nut *b* was adjusted to give spring *a* a tension of 100 pounds for each adjustment of capscrew *c*.

Although the results of this study as given below are believed to be worth while and to give a rather definite idea of the operating characteristics of these hitches, the following limitations should be noted:

1. No attempt was made to study all the spring overload release hitches on the market.

2. Not enough tests were made to determine the effect of wear resulting from long-continued use.

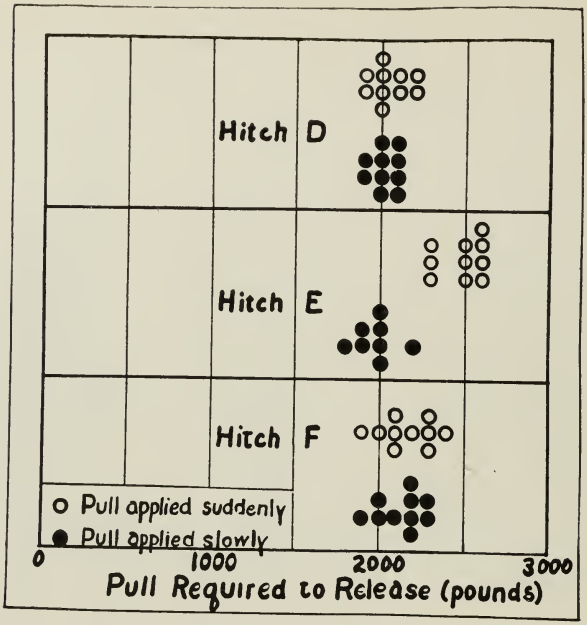

Fig. 16.—Note the uniformity of the results of tests at the same setting.

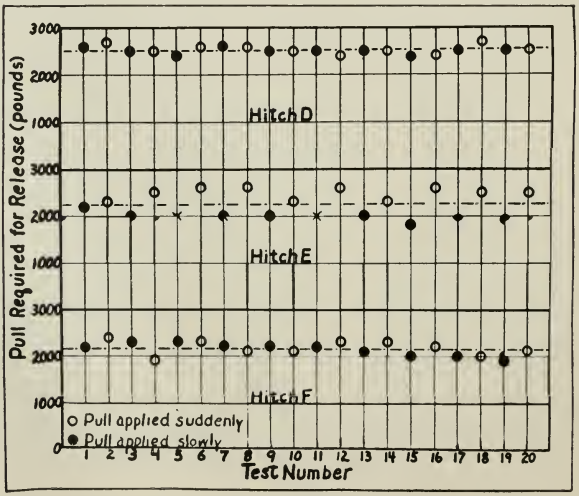

Fig. 17.—Note effect of wear resulting from use, and effect of speed of application of pull, on pull required to trip release when set at a given adjustment. In tests 5 and 11 of hitch *E*, the hitch tripped, but the hook caught on the clevis so that the load was not released until the pull was over 4,000 pounds; this is indicated by the two stars in the figure.

3. No attempt was made to determine the effect of long periods of weathering.

4. While the elongation of the hitch resulting from the use of the shock dynamometer was only 0.05-inch per 1000 pounds of pull, conditions under a suddenly applied pull were not quite the same as they would have been without the dynamometer in the hitch.

5. The dynamometer was read only to the closest 100 pounds.

Tests at Uniformly Varied Adjustments.—The tests, results of which are shown in figure 15, were made to determine the approximate relation between uniform changes in adjustment and the pull required for release. The change in adjustment was, for hitch *D*, .01-inch in the relative location of latch *c;* for hitch *E*, one-half turn of cap-screw *c;* and for hitch *F*, two and three-quarters turns of nut *b* (figs. 13 and 14). No change was made in the adjustment of nut *b* in hitch *D*, because this would have had practically the same effect as a corresponding change in the location of latch *c*. Two additional series of tests, also run with different adjustments of nut *b* of hitch *E*, are explained in connection with figure 15.

TABLE 2

Result of Twenty Tests at the Same Adjustment*

	Hitch D	Hitch E	Hitch F
	pounds	*pounds*	*pounds*
Maximum	2,700	2,600	2,400
Minimum	2,400	1,800	1,900
Average of slowly applied pulls	2,510	1,975	2,140
Average of suddenly applied pulls	2,540	2,420	2,170
Average of all tests	2,525	2,255	2,155
Average of first four tests	2,575	2,250	2,200
Average of last four tests	2,575	2,200	2,000

* Slow-steady and sudden-jerky applications of the drawbar pull were alternated during these tests.

Tests at Constant Adjustments.—To give an idea of the uniformity of the pull required for release at a given adjustment, the effect of speed of applying the pull, and the effect of wear resulting from use, twenty tests at the same adjustment, in which slow and sudden applications of the drawbar pull were alternated, were run on each hitch. The results are shown in table 2, and in figures 15, 16, and 17. These results are much more uniform and consistent than those obtained with wooden breakpins.

Conclusions.—Considered as a group, the hitches studied included most of the desirable characteristics of the ideal spring overload re-

lease hitch. There appears, furthermore, to be no fundamental mechanical reason why almost all of these desirable characteristics can not ultimately be combined into a single spring overload release hitch. Where there is actual need for overload protection, such a hitch, if properly cared for and adjusted, should give more satisfactory performance than breakpins.

ACKNOWLEDGMENTS

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